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Investigation of Centrifugal Pump Used in Irrigation by Numerical (CFD Analysis) and Experimental Methods

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Sulamada Kullanılan Santrifüj Pompanın Sayısal (HAD Analizi) ve Deneysel Yöntemlerle İncelenmesi

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Abstract
The goal in centrifugal pump designs is to achieve maximum efficiency. Even small percentage increases in pump efficiency provide high overall energy savings in pumps that operate for long periods of time, especially those that use high-power motors. While fan and casing structures are designed in line with this goal, many structural features affect the efficiency of the pumps. In the study, pump performance values of the 5" horizontal shaft centrifugal pump at different flow rates were measured in the laboratory. Depending on the change in pump flow rates, it was determined that the pump power consumption varied between 15-42kWh and the efficiency values varied between 32.63-55.57%. Considering the statistical compatibility of the measured pump performance values with the values obtained by CFD, it was determined that the coefficient of determination of the measured and predicted values varied between 0.870 and 0.936, which is quite high. The mean absolute percentage error was determined to be well below the acceptable value of 10%.

Keywords: *Centrifugal pump, power consumption, manometric height, efficiency, CFD*

Öz
Santrifüj pompa tasarımlarında amaç maksimum verime ulaşmaktır. Pompa verimliliğindeki küçük yüzdesel artışlar bile, özellikle yüksek güçlü motorlar kullanan, uzun süre çalışan pompalarda yüksek toplam enerji tasarrufu sağlar. Fan ve gövde yapıları bu amaç doğrultusunda tasarlanırken birçok yapısal özellik pompaların verimini etkilemektedir. Çalışmada 5"'lik yatay milli santrifüj pompanın farklı debilerdeki pompa performans değerleri ölçülmüştür. Pompa debilerindeki değişime bağlı olarak pompa güç tüketiminin 15-42kWh arasında değiştiği, verim değerlerinin ise %32,63-55,57 arasında değiştiği tespit edilmiştir. Ölçülen pompa performans değerlerinin CFD ile elde edilen değerlerle istatistiksel uyumu göz önüne alındığında, ölçülen ve tahmin edilen değerlerin determinasyon katsayısının oldukça yüksek bir değer olan 0,870 ile 0,936 arasında değiştiği saptanmıştır. Ortalama mutlak yüzde hatasının ise kabul edilebilir sınır değer olan %10'un oldukça altında olduğu belirlenmiştir.

Anahtar Kelimeler: *Santrifüj pompa, güç tüketimi, manometrik yükseklik, verim, HAD*

1. Introduction

Irrigation in agriculture is the delivery of water needed by the plant, which cannot be met by rainfall, to the root zone of the plant in the soil, at the required place and time. It has been determined that 57 billion m3 of our country's total water potential is used for various purposes, 44 billion m3 (77%) of which is used as irrigation water and 13 billion m3 (23%) is used as drinking-use and industrial water. Our country has 24 million hectares of arable agricultural land, and the economically irrigable amount of this is determined as 8.5 million hectares. In our country, approximately 81.9% of the 8.5 million hectares of economically irrigable agricultural area can currently be irrigated. Turkey's population is announced as 84,680,273 people by TURKSTAT as of 2022. While the annual amount of water per capita in our country was 1 652 m3 in 2000, it decreased to 1 322 m3 in 2022. Considering the usable water potential per capita, Turkey is among the countries experiencing water stress. Therefore, it is important to use water economically and optimally.

Design parameters such as number of blades, blade exit angle, blade width and impeller exit diameter affect pump performance and energy consumption. Impellers with intermediate blades are used to eliminate the low efficiency, which is one of the biggest problems of low specific speed pumps, and to ensure more efficient use of energy. By adding intermediate vanes at rates of 40, 55,

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70 and 85% of the main vane length to submersible pump impellers with different blade numbers and blade exit angles, the effects of the intermediate vane length on the submersible pump performance were examined. The optimum number of blades was found to be z=6, the optimum blade exit angle 25° and the optimum intermediate blade length was 0.70·L ~ 0.85·L (Korkmaz, 2015).

The effect of impeller design variables on efficiency and cavitation performance in centrifugal pumps with different specific speeds is important. In this study conducted by Demirel (2021), impeller design was made with CFTurbo software for centrifugal pumps with four different specific speeds, and efficiency and cavitation performances were determined by performing Computational Fluid Dynamics (CFD) analysis of the designed impellers.

Calisir et al. (2003) calculated the performance values of centrifugal pumps in their study with the help of statistical prediction equations and Artificial Neural Network. In training ANNs, the extended delta-bar-delta learning algorithm with a multilayer perceptron structure was used as data for the experimental results and measured structural parameters of 21 different pumps in the training and testing process. They stated that the results obtained from the neural model were closer to the experimental data than the results obtained from the statistical prediction equations.

Taner (2007) designed centrifugal pumps using ANN techniques. For this purpose, the pump speed, flow rate and manometric height parameters were used as input data, and the impeller inlet diameter, impeller outlet diameter, number of impeller blades, pump inlet pipe diameter and pump outlet pipe diameter parameters were used as output data. The mean error in prediction to the measurement values of the pump design parameters, impeller inlet diameter, impeller outlet diameter, number of impeller blades, pump inlet pipe diameter and pump outlet pipe diameter, was 2.43%, 3.37%, 2.11%, 3.29% and 5.11%, respectively. The degree of agreement was obtained as 0.0009, 0.0008, 0.0007, 0.0012 and 0.0019, respectively.

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Toprak (2007) tested vertical shaft deep well pumps with three different nominal diameters (4”, 5” and 6”) at optimum speed and 6 different number of stages and investigated the effect of the number of stages on pump efficiency and efficiency components. According to the research results, increasing the number of stages increased the overall efficiency of the pump. The increase in efficiency is caused by the fact that the increase in the power absorbed by the pump due to the increase in the number of stages is smaller than the increase in the flow rate and pressure values produced by the pump.

The use of computational fluid dynamics in pump designs provides advantages in terms of time and cost by reducing repetitive prototyping and testing processes, and makes it possible to obtain data on the flow in the pump even at points that are impossible to observe experimentally. Thus, if flow characteristics that may cause a decrease in efficiency are observed, corrections can be made to the pump geometry. In the study, the pump performance curve obtained from the analysis for five flow rates was compared with the pump performance curve obtained from the tests. When these five points were compared with the tests, it was seen that the error rate did not exceed 10% (Uçar, 2017). The 3-stage centrifugal pump, whose design parameters are 2850 rpm, 12.5m3h-1 flow rate and 40 meters head, has been tested at 1900 rpm operating conditions as well as the design cycle. The flow structure between the two blades within the pump impeller was determined experimentally in the middle of the impeller exit width using the Particle Imaging Velocimetry (PIV) method. The relative difference between the PIV-CFD results obtained with 16 different points created within the pump impeller was found to be between 5.7-21.2% (Aksoy, 2018).

In this study, the operating characteristics (manometric height, power consumption and efficiency) of a locally made, single-inlet, stepless, horizontal shaft centrifugal pump with a nominal diameter of 5", which is widely used in irrigation, were determined at 10 different openings of the discharge line control valve. The performance output of the pump obtained through CFD analysis was compared with experimental results.

2. Materials and Methods

Centrifugal pump, S.Ü. It was tested in the Pump Test Unit of the Faculty of Agriculture, Department of Agricultural Machinery and Technologies Engineering. The general view of the pump test unit is given in Figure 1. The water tank (1), which has a volume of approximately 50 m3, does not contain solids, is cold and clean, and is 4 m deep. The suction line is equipped with a foot valve and strainer (2), a suction pipe (3), a 90 mm suction line elbow (4), a suction line adjustment (control) valve (5) and a vacuum meter (6). The discharge line is equipped with a vertical outlet pipe (7), a 90 mm elbow (8), a horizontal discharge line pipe (9), a pressure gauge (10) and a discharge line adjustment (control) valve (11). Movement to the pump (12) is transmitted from the mechanical variator (14) coupled to the electric motor (13) with the help of a jointed shaft (15). There is a control panel with electrical indicators in section 16. Some technical specifications of the tested pump are given in Table 1. The dimensions of the physical dimensions of the pump given in Table 1 are given in Figure 2.

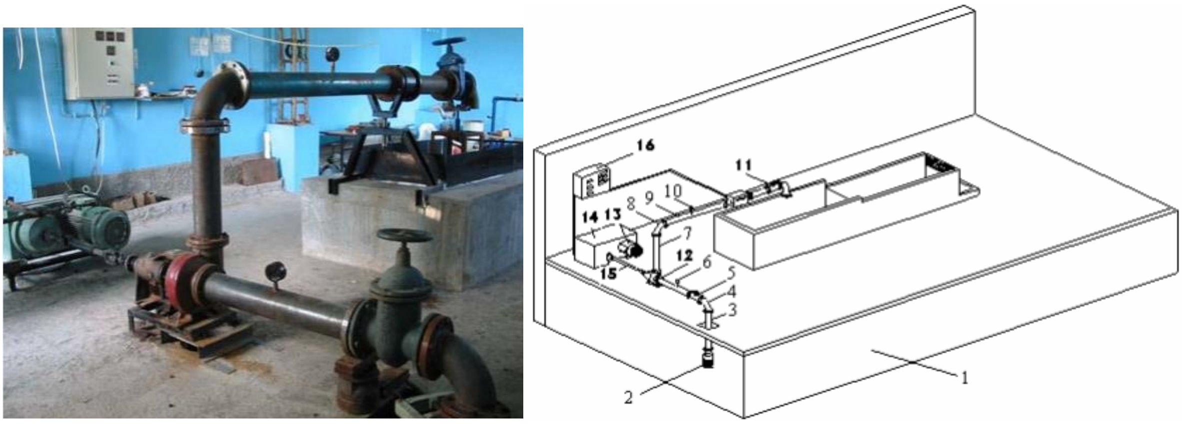

Figure 1. Experimental unit where pump trials were carried out

Table 1. Some technical specifications of the pumps used in the trial.

Pump (“)	D_1 (mm)	D_2 (mm)	b_2 (mm)	z (adet)	D_e (mm)	D_b (m)
5	140	260	27	8	125	120

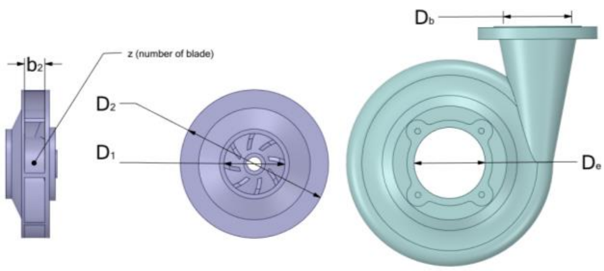

Figure 2. Schematic view of pump dimensions

The symbols in the pump dimensions De indicates the pump suction diameter, Db indicates the discharge diameter, D1 indicates the pump impeller suction diameter, D2 indicates the pump impeller outer diameter. The discharge width of the impeller is expressed by b_2 and the number of vanes is expressed by z. First, a manometer with a measurement range of 0-25 kgcm^{-2} was connected to the discharge line. Then, the suction line control valve was fully opened and the discharge line control valve was fully closed. The output speed setting of the mechanical variator was minimized and the engine was given initial start. The variator speed was increased to the prescribed number of revolutions and the highest manometer pressure was read. The flow rate, positive and negative pressures, the electrical power drawn by the system from the network and the data of the experimental unit at 10 valve openings, starting from the fully closed position of the suction line control valve at the set speed, until the discharge line control valve is fully open. Static suction height, diving depth and vertical height values between manometer and vacuum meter were recorded.

In centrifugal pump trials, output flow values were measured volumetrically with an electromagnetic flowmeter. Electromagnetic flowmeters with flanged digital displays, resistant to 16 bar pressure, were used in accordance with the column pipe diameter of the pump. The outlet pressure was in the 0-10 bar pressure range, and the pressure values were determined with the help of a manometer. Pressure values were used to obtain manometric height. In the pump, manometric height (Hm) was calculated with the following equation (Baysal 1975, Tezer 1978).

$$Hm = 10.2Pb + 0.0136Pe + Hk + Hv + \Delta z \quad (1)$$

Here; Hk is the pipeline friction losses (m), Hv is the velocity energy height of the pumped water (m), Δz is the vertical distance between the monometer-vacuummeter axes (m).

The measurement of the power drawn from the speed was made with a power analyzer. Suitable operating conditions were ensured by setting the pump operating speed to 2350 min^{-1} with the help of the electronic panel. Electric power was measured as the power drawn from the mains (N_s) with the help of the panel.

The following links are given for pump properties and performance indicators to be obtained from experimental and numerical results (Babayiğit et al. 2015).

$$N_h = \frac{\rho g Q H_m}{1000} \quad (2)$$

$$N_m = \frac{T_S.\omega}{1000} \quad (3)$$

Among the symbols in the equations, N_h indicates hydraulic power, ρ indicates fluid density, g indicates gravitational acceleration, Q indicates volumetric flow rate, and Hm indicates head. All units comply with the SI unit system and basic units are used. The reason for dividing the power by 1000 is to show the power in kWh. On the other hand, N_m indicates shaft power, T_s indicates the torque in numerical analysis, and ω indicates angular speed. Pump efficiency is calculated by considering the ratio of pump hydraulic power to shaft power.

The predictive ability of the CFD was examined according to the statistical methods. In order to determine the performances of the results, ε, RMSE, R^2 and D values that are considered to be the principal accuracy measures and that are based on the concept of the mean error and are commonly used were calculated using the following equations (Willmott et al. 1985; Kashaninejad et al., 2009; Çarman and Taner, 2012):

$$\bar{\varepsilon} = MAPE = \frac{100\%}{n}\sum_{i=1}^{n}\left|\frac{y_i - \tilde{y}_i}{y_i}\right| \quad (4)$$

$$\mathrm{RMSE} = \left(\frac{1}{n}\sum_{i=1}^{n}(y_i - \tilde{y}_i)^2\right)^{1/2} \quad (5)$$

$$R^2 = 1 - \left(\sum_{i=1}^{n}(y_i - \tilde{y}_i)^2\right) / \left(\sum_{i=1}^{n}(y_i - \bar{y})^2\right) \quad (6)$$

$$D = 1 - \sum_{i=1}^{n}(y_i - \tilde{y}_i)^2 \,/\, \sum_{i=1}^{n}[|y_i - \bar{y}| + |\tilde{y}_i - \bar{y}|]^2 \quad (7)$$

Where RMSE is the root-mean-square error, R^2 is the coefficient of determination, ε is the mean absolute percentage error, D is the agreement index of the system, n is the number of data, y_i is the measured value, $\tilde{y}$ is the predicted value, and $\bar{y}$ is the mean value.

With the developments in today's technologies, some programs and software have been developed to determine pump performance. ANSYS FLUENT numerical analysis program is one of these software. "SpaceClaim", one of the modules in the program, was used to create pump stage models. The pump used in the experiments was designed according to the size parameters and the flow volume was created according to the pump geometry and its flow was defined as water. The flow volume was divided into finite volumes with the help of ANSYS Mesh to make the created pump model ready for analysis. Figure 3 shows the pump flow volume and the structure of the mesh.

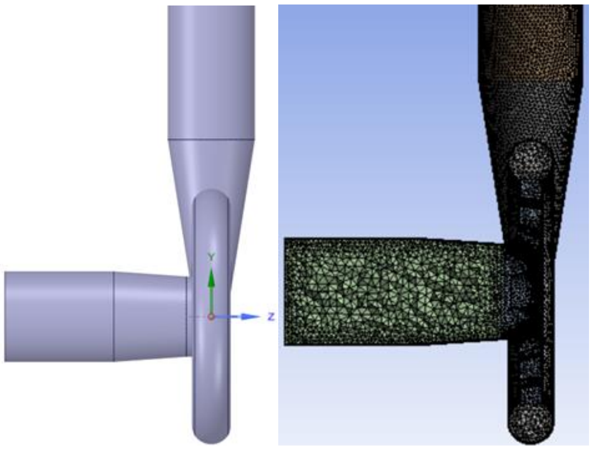

Figure 3. Pump flow volume and mesh structure

In the finite volume method, experiments were carried out assuming that the results did not change up to a certain mesh number, and were considered independent of the mesh. Numerical results were obtained by changing the geometric and operating parameters in the analysis after the mesh structure remained unchanged. Figure 4 shows the variation of mesh number with pump outlet pressure. As a result of the analysis, it was determined that the number of mesh was sufficient at the points where constant outlet pressure values were obtained (Xiong et. al, 2023).

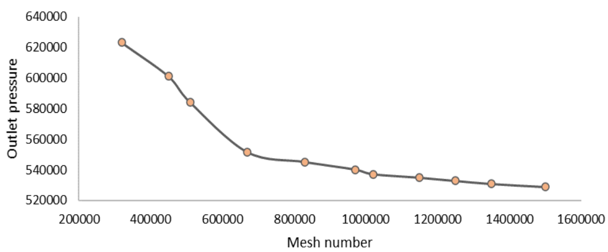

Figure 4. Suitability of mesh structure

FLUENT software is included in the program modules as computational fluid dynamics (CFD) analysis software that works according to the finite volume method. Analyzes were carried out with this module, depending on the geometry, for the nodes located in the solution area divided into finite volumes. Analyzes can be made with multiple turbulence models in the FLUENT module. The most commonly used method for pump solution in these models is the k-ϵ turbulence model, which was used for numerical analysis (Ding et al.2019; Wang et. al, 2023). Establishing the necessary boundary conditions is an important parameter for the suitability of the solution. In the study, analyzes were carried out by accepting pressure at the inlet and mass flow at the outlet as the boundary conditions in the pump CFD analyses.

In determining the pump performance, Caridad et al. (2004) determined the CFD inlet and outlet conditions as pressure inlet and mass flow outlet. In another study, researchers set boundary conditions for pump numerical analyses. These conditions were total pressure for inlet and mass flow rate for outlet (Khoeini et al. 2019). Wang et al. (2023) defined the inlet and outlet conditions for the pump performance as total inlet pressure and mass flow rate for the outlet in numerical studies. They also preferred the k-ε model for the turbulence model.

In CFD analyses, previous studies were used for the suitability of inlet and outlet conditions. Different inlet and outlet conditions can be defined in the literature. When determining the boundary conditions, the boundaries of the gauge pressure for the inlet and mass flow output for the outlet were determined considering the previous studies. The boundary conditions obtained for the CFD analysis of the tested pump are given in Table 2.

Table 2. Pump geometry and inlet-outlet boundary conditions for CFD analysis

Analysis Conditions	Values
Fluid type/Solution model	Water / k-ε Realizable
Number of Elements	1400000-1500000
Element conformity value (Skewness)	0.89
Number of Transfer	2350 min^{-1}
Inlet condition (Pressure inlet)	0 Pa
Output condition (Mass flow outlet)	(8.33/13.89/19.44/22.22/25/27.78/30.55/33.33/38.89/44.44) kg s^{-1}

3. Results and Discussions

During the trials, the ambient temperature varied between 15-29°C, the relative humidity varied between 51-72%, and the pumped water temperature varied between 15-17°C. The test results of the pump at different valve openings in the discharge line are given in Table 3. The manometric height values of the pump varied between 50-60.5 m and the power requirement varied between 15-42 kWh. An approximately 4-fold increase in pump flow rate increased the power requirement by 1.8 times.

Depending on the change in flow rate, pump efficiency values varied between 32.63-55.57%. Depending on the increasing flow rate, it is seen that the power values increase while the manometric height values decrease. When the relationship between flow rate values and efficiency is examined, it is observed that while efficiency increases with increasing flow rate values, efficiency decreases after a certain flow rate value. This can be attributed to the increase in turbulence losses as a result of operating outside the design flow rate and the inability to achieve Reynolds similarity. (Baysal 1975, Tezer 1978, Karassik ve ark. 1986, Stepanoff 1993, Yazıcı 1996).

The experimental results obtained from these studies are given in Table 3. The optimum efficiency was determined as a result of the pump performance values. At the determined efficiency point, the head was 54 m and the power drawn was 37 kWh. Experimental and numerical analysis results of the pump used in the study were compared. In this comparison, the changes in pump pressure, power and efficiency at different flow rates were examined. It is seen that the structural properties and behavior of the fluid in the pump, which is simulated with the numerical analysis method, changes.

Table 3. Performance values obtained at 10 different openings of the discharge line valve.

Q ($m^3 h^{-1}$)	Hm (m)	Np (kWh)	η (%)
30	60	15	32.63
50	60.5	17,5	47.01
70	60	23	49.66
80	59	25	51.35
90	58.5	29	49.37
100	58	31	50.88
110	57	33	51.67
120	56	35	52.22
140	54	37	55.57
160	50	42	51.80

When the pump numerical results are analysed the pressure changes are given in figure 5. When the pressure contours obtained in the pump pressure line were examined, it was found that the pressure values were not equally distributed at all flow rates except for the separation zones where the velocity changed significantly.

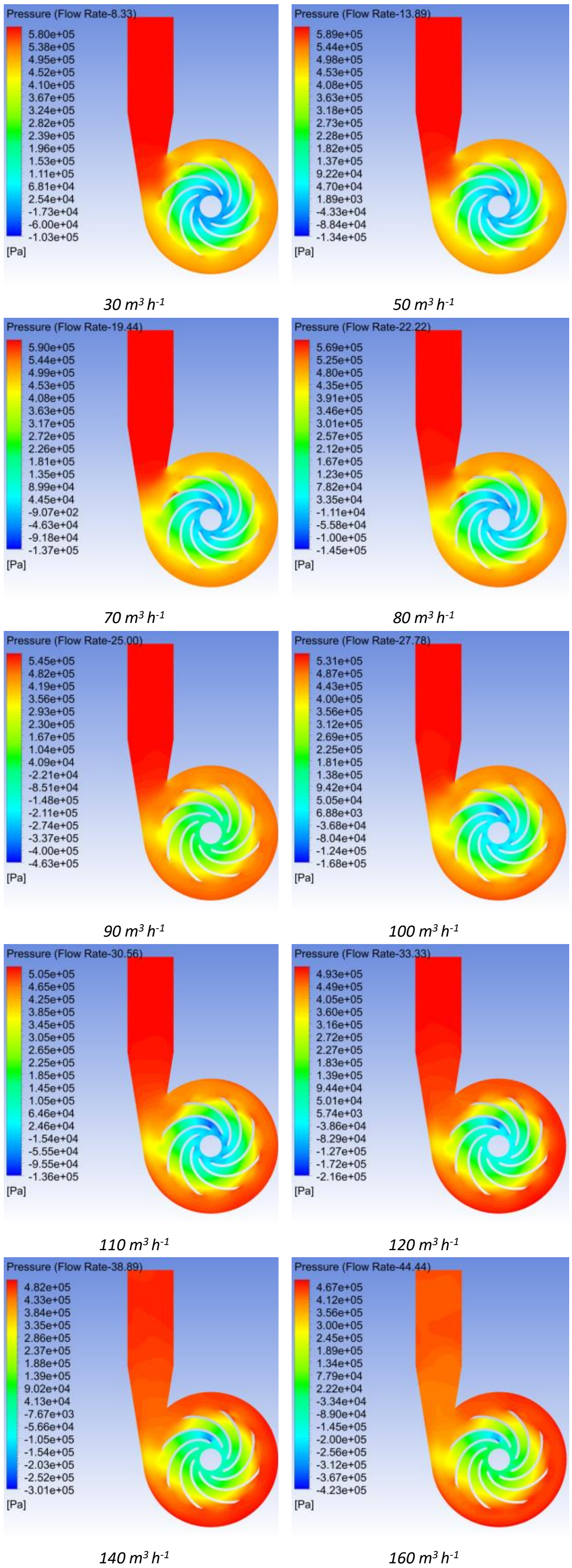

Figure 5. Pressure distributions of the centrifugal pump at different flow rates.

The lowest point of the pressure values was realised at the suction line for each flow rate value. here the pressure corresponds to negative values. in general, at the points where the fluid leaves the pump vanes, it is seen that the pressure values are at a low level due to the centrifugal body structure, while it is determined that the pressure rises even more at the walls. When the pump and experimental results are compared, the pump outlet pressure is found to be approximately 590000 Pa at a flow rate of 70 m^3h^{-1}. At the best efficiency point; the maximum pressure value at 140 m^3h^{-1} (38.89 kgs^{-1}) is approximately 482000 Pa.

As a result of the numerical analysis of the pump, it is observed that the pressure and velocity contour changes from low flow rate to high flow rate (Figure 6). When the velocity distribution at low flow rates is analysed, the distribution of the contours shows that there is backflow with the fluid in the pump. Due to this backflow, higher velocity contours are formed in and around the impeller. Considering the point where the efficiency is the highest, it can be said that the velocity contours move towards the pump outlet, thus reducing the backflow in the pump. It is also observed that the pressure decreases with increasing speed in the pump in both experiments. When we consider the pump speeds, it is seen that the speeds may change at some points locally. in general, the changes in the speeds formed in and around the pump impeller differed at each flow rate value. this can be explained by the turbulence and backflow in the region where the pump leaves the impeller. When the results are examined, it is seen that the highest speed is approximately 45.20 ms^{-1} at maximum flow rate. at the point where the efficiency is the highest, the speed is 39. 99 $ms^{-1.}$ The predictive ability of CFD is investigated according to mathematical and statistical methods. Table 4 shows the mean values of RMSE, R^2, D and Ɛ of the CFD to predict each of the three output parameters.

Table 4. Goodness-of-fit statistics of pump characteristic values predicted by CFD.

	ε(%)	R^2	RMSE	D
H_m(m)	4.575	0.870	2.894	0.851
N(kWh)	3.229	0.935	2.084	0.983
η(%)	1.163	0.936	1.492	0.986

The mean absolute percentage error ranged from 1.163 to 4.575 %. The smallest error value of 1.163 % was obtained in predicting the pump efficiency. The mean absolute percentage error was found to be well below the acceptable value of 10 % (Taner, 2007). The statistical values RMSE was found as 2.894 for Hm, 2.084 for Np, and 1.492 for η. It has been calculated that the coefficient of determination used in the prediction of pump performance characteristics varies between 0.870 and 0.936, and the agreement index varies between 0.851 and 0.986.

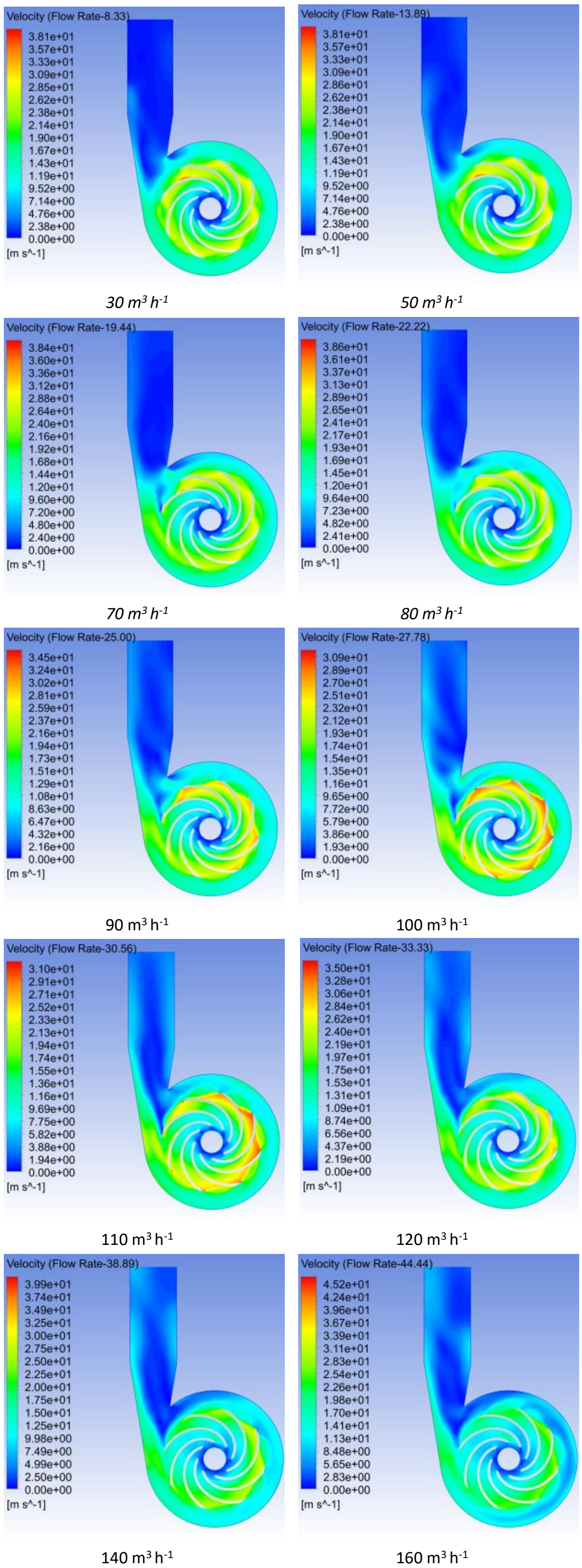

Figure 6. Pressure distributions of the centrifugal pump at different flow rates.

The fact that the value of both criteria is close to one is an indicator of the success of the CFD prediction results. Mercan (2018) developed an prediction method for calculating the performance of a centrifugal pump using loss correlations. When compared to experimental and CFD results of centrifugal pumps, the average absolute error of manometric height values was found to be 26.9 %. Uçar (2017) analyzed the flow in the impeller and volute of a centrifugal pump with the Ansys CFX program. Resercher compared the manometric height values obtained at 5 different flow rates ranging from 4 to 12 m^3h^{-1} with the values obtained by CFD.

According to the test results of CFD results, the error rates were below 10 % at all five points. The flow structure obtained by Computational Fluid Dynamics was compared with the experimental PIV results. The difference between the pump characteristic values obtained in CFD and experimental studies was found to be in the range of 1-9% (Aksoy, 2018).

The correlation coefficient (R) of the linear relationship between the experimental data and the predicted values obtained from CFD was found to be 0.956, 0.998 and 0.993 for H_m (m), N_p (kWh) and η(%), respectively (Figures 7, 8 and 9).

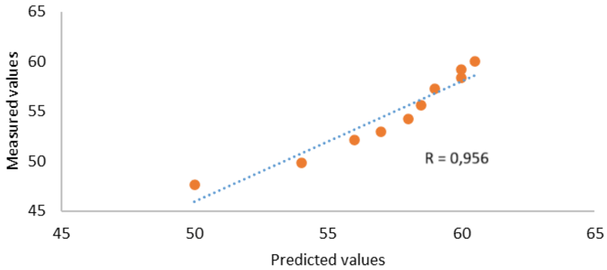

Figure 7. The relationship between measured and predicted values of manometric height.

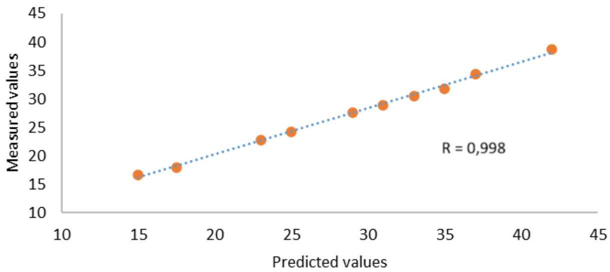

Figure 8. The relationship between measured and predicted values of pump shaft power requirement.

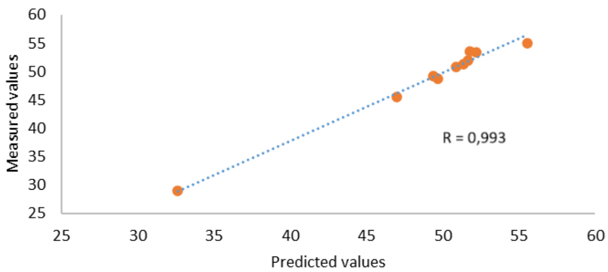

Figure 9. Relationship between measured and predicted values of pump efficiency.

4.Conclusions

In the study, performance values were obtained at 10 different openings of the discharge line valve at a constant speed of the pump shaft. Depending on the change in flow rate, pump efficiency values varied between 32.63-55.57 %. A parabolic relationship was observed between pump flow rate and efficiency values.

When the velocity distribution at low flow rates is examined in CFD analysis, the distribution of the contours shows that there is backflow with the fluid in the pump. Due to this backflow, higher velocity contours have formed in and around the impeller. Considering the point where the efficiency is highest, it can be said that the speed curves move towards the pump outlet, therefore the backflow in the pump decreases.

Also in the study, CFD was analyzed for the prediction of the performance parameters of centrifugal pump. The overall results show that the CFD can be used as an alternative method to find the performance parameters in these systems. The difference between the average of the measured and predicted values of the pump was determined as 4.5% for manometric height, 4.9% for power consumption and 0.7% for efficiency. In addition to its numerical accuracy, the CFD analises is much faster and easier to use, which makes it suitable for generating the performance parameters of pump. Such analyzes can be used as a reference for future pump studies.

Declaration of Ethical Standards
The authors declare that they comply with all ethical standards

Credit Authorship Contribution Statement
Author 1: Investigation, Methodology, Experimental study, Writing, Study design
Author 2: Investigation, Methodology, Experimental study, Writing, Statistical analysis

Declaration of Competing Interest
The authors have no conflicts of interest to declare regarding the content of this article.

Data Availability Statement
All data generated or analyzed during this study are included in this published paper.

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